MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 3, 2018/2019

EME3236 – HEATING, VENTILATION AND AIR CONDITIONING SYSTEMS

(ME)

27 MAY 2019
09.00 a.m. - 11.00 a.m.
(2 Hours)

INSTRUCTIONS TO STUDENTS

1. This question paper consists of seven pages (including the cover page) with four questions and an Appendix.

2. Answer ALL four questions.

3. Each question carries 25 marks and the distribution of the marks for each question is given in brackets [ ].

4. Write all your answers in the answer booklet provided.

5. Property-tables booklet is provided for your reference.

Question 1

Consider the moist air enters the adiabatic fan coil unit at the rate of 100.5 g/s as shown in Figure Q1. The moist air temperature and absolute humidity at the inlet (State 1) are 5 °C and 0.005 respectively. The moist air is heated to 40 °C (State 2). Then, it is mixed with spray liquid water at 19 °C (Close to the wet-bulb temperature at 40 °C and absolute humidity of 0.005) and assumes all spray liquid water is evaporated. The conditioned air is delivered to match the human comfort condition at 23 °C (State 3). The flow throughout the fan coil unit is maintained at atmospheric pressure, p_T = 101.42 kPa. Assume the constant pressure specific heats of dry air, c_{pa} = 1.005 kJ/kg·K and water vapor, c_{pv} = 1.8723 kJ/kg·K.

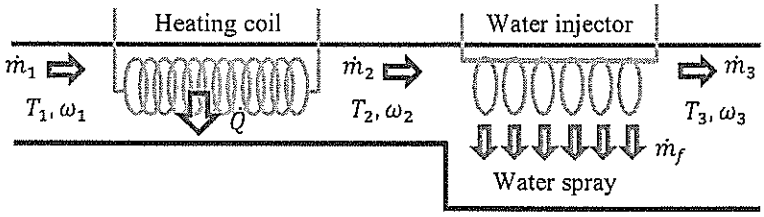

Figure Q1

a. Calculate the relative humidity at the inlet.

[2 marks]

b. Calculate the relative humidity after heating.

[2 marks]

c. Determine the rate of heat transfer from the heating coil.

[2 marks]

d. Determine the rate of liquid water injection.

[6 marks]

e. Find relative humidity at the outlet.

[2 marks]

If the liquid water is sprayed at 25 °C, and the conditioned air is delivered at the temperature of 30 °C (State 4).

f. Determine the rate of liquid water injection.

[5 marks]

g. Find relative humidity at the outlet.

[3 marks]

h. Sketch and label state 1, 2, 3 and 4 on a psychrometric chart.

[3 marks]

Continued...

Question 2

A (3 m x 3 m) horizontal glass is installed on the roof of a factory as shown in Figure Q2. The single layer glass has 5 mm thickness with thermal conductivity, k = 0.78 W/m·°C. At a night with clear sky, the indoor air temperature of the factory is kept at 20 °C while the outdoor air temperature is 5 °C. Both indoor and outdoor airs are quiescent. The internal glass surface temperature is 10 °C and the effective sky temperature is -30 °C.

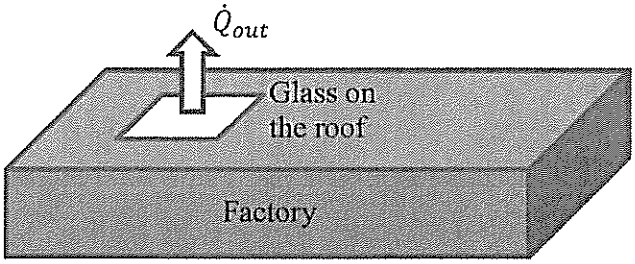

Figure Q2

Neglect the radiation heat transfer inside the factory and assume ideal gas for air where the thermal expansivity is equal to T^{-1}.

a. Find the characteristic length and Prandtl number of convection heat transfer for the horizontal glass.

[2 marks]

b. Find the Rayleigh number of convection heat transfer for the internal glass surface.

[3 marks]

c. Calculate the rate of heat loss through the glass.

[6 marks]

d. Determine the temperature of the external glass surface.

[3 marks]

e. Find the Rayleigh number of convection heat transfer for the external glass surface.

[3 marks]

f. Determine the emissivity of the glass.

[8 marks]

Continued...

Question 3

In a particular section of a hotel, the layout of the floor plan with the information of air conditioning ducting network is shown in Figure Q3. The ducting is running through the center of the corridor. There are dampers placed at the tee of the duct to block the air supply into the rooms, which are not occupied. At the diffuser of each room, volume flow rate of air supply is controlled so that it is evenly distributed to all the rooms at 30 liter/s. The duct diameter of the main and the branch network are $D_m = 0.5$ m and $D_b = 0.2$ m respectively. As the thickness of partition walls for every room is rather small as compared to the room sizing, it is neglected in the floor plan. The surface roughness of the pipe is 0.045 mm. The minor lost coefficients, K_M at elbow is 0.5 and for tee, it is shown in the following table with respect to the ratio of volume flow rate , $\dot{\forall}_b/\dot{\forall}_m$, and diameter, D_b/D_m, of branch duct to main duct.

$\frac{D_b}{D_m}$	$\frac{\dot{\forall}_b}{\dot{\forall}_m}$	K_M	
		Main	Branch
0.4	1/8	6.0	13
	1/7	3.0	9.0
	1/6	2.0	6.0
	1/5	1.0	5.0
	1/4	0.5	3.0
	1/3	0.2	1.5
	1/2	0.1	0.8
1.0	1/3	1.0	1.0
	1/2	0.5	0.5

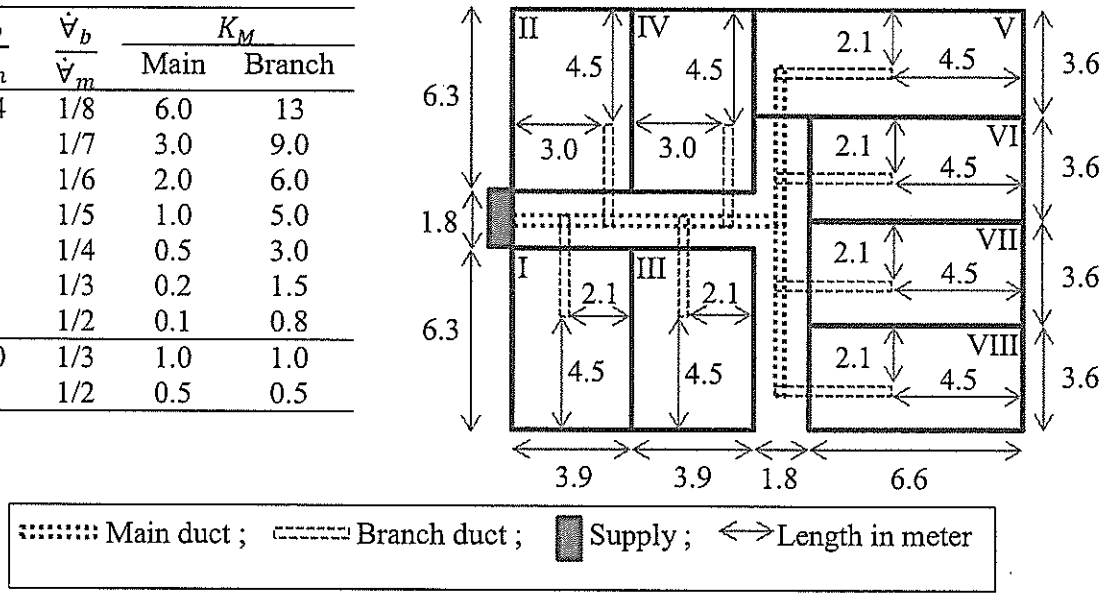

Figure Q3

If room III, V, and VIII are occupied and one of the dampers is fully open where the local head loss is assumed zero, using Haaland formula for duct friction to evaluate the following.

a. Determine the total head loss of the ducting system (excluding the losses at the dampers of the diffusers).

[13 marks]

b. Calculate the head loss to each room.

[9 marks]

c. Determine and the head loss of each diffuser.

[3 marks]

Continued...

Question 4

Two separate water pumps are used to deliver the cold water from the evaporator of the chiller to the air handling units (AHU) located at third and fifth floor of the building as shown in Figure Q4. The AHU takes the air outside the building and delivers the fresh air into the particular floor. The water pipe diameter in network 1 is 0.02 m. The volume flow rate of pipe network 1 is 1 liter/s. The pipe wall has smooth surface. Both pumps 1 and 2 have identical and constant pressure head even though they are delivering different volume flow rate. The kinematic viscosity of water is 1.4 $\times 10^{-6}$ m^2/s.

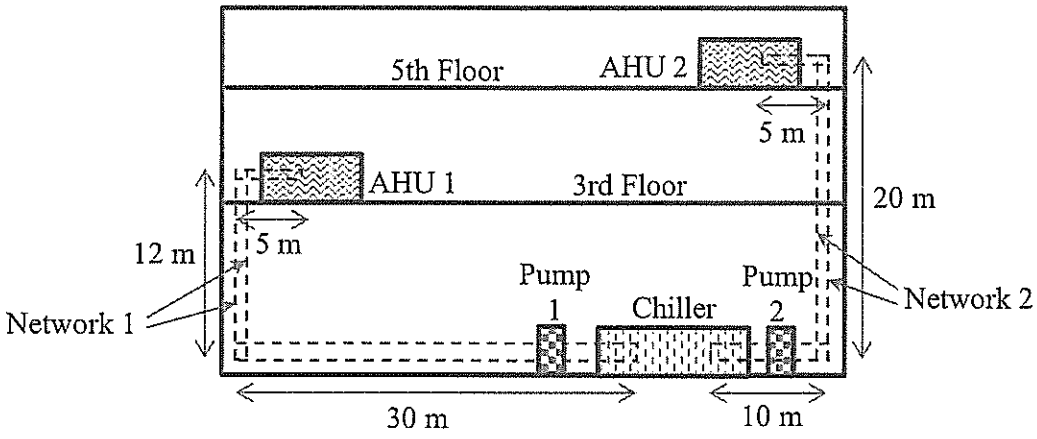

Figure Q4

As the pipes are sufficiently long, the minor loss at the elbows can be neglected.

a. Find the total head loss of network 1.

[8 marks]

b. Calculate the power consumption of the pump 1 if the efficiency is 80%.

[3 marks]

c. If the volume flow rate of network 2 is equal to the volume flow rate of network 1, determine the pipe diameter and the corresponding Reynolds number of network 2.

[8 marks]

d. If both networks have the same pipe diameter, evaluate the volume flow rate and average flow velocity of pipe network 2.

[6 marks]

Continued...

APPENDIX A : FORMULA AND CONSTANTS

A1. Properties of air and water: density, thermal conductivity, heat capacity, kinematic viscosity
Air at 25 °C: ρ_a = 1.23 kg/m^3, k_a = 0.0246 W/m·°C, c_{pa} = 1.005 kJ/kg·K, ν_a = 1.46 × 10^{-5} m^2/s
Water at 25 °C: ρ_w = 1000 kg/m^3, k_w = 0.598 W/m·°C, c_{pw} = 4.18 kJ/kg·K, ν_w = 1 × 10^{-6} m^2/s
Water vapor: c_{pv} = 1.872 kJ/kg·K.

A2. Physical constants
Stefan-Boltzmann constant, σ = 5.67 × 10^{-8} W/m^2·K^4, Universal gas constant, R = 8.314 kJ/kmol·K
Gravity acceleration, g = 9.81 m/s^2, Standard atmospheric pressure, p_0 = 101.325 kPa

A3. Relation between relative humidity, absolute humidity, dry-bulb and wet-bulb temperature:

$$\phi = \frac{\omega}{\omega + 0.622}\left(\frac{p_T}{p_{v*}}\right) \; ; \; \omega = \frac{0.62\phi p_{v*}}{p_T - \phi p_{v*}} \; ; \; \omega = \frac{\left(\frac{0.622}{p_T/p_{v*@Twb} - 1}\right) h_{fg@Twb} - c_{pa}(T_{db} - T_{wb})}{h_{fg@Twb} + c_{pv}(T_{db} - T_{wb})}$$

A4. Friction Factor in term of Reynolds number and Roughness ratio

Laminar (Re < 2000)

$$f = \frac{64}{Re}$$

Transition (Blasius - Smooth)

$$f = 0.316(Re)^{-\frac{1}{4}}$$

Fully Turbulent ($Re \gg 2000$)

$$\frac{1}{\sqrt{f}} = -2.0 \log\left(\frac{\epsilon/D}{3.7}\right)$$

Transition (Haaland - Explicit)

$$\frac{1}{\sqrt{f}} = -1.8 \log\left[\left(\frac{\epsilon/D}{3.7}\right)^{1.1} + \frac{6.9}{Re}\right]$$

Transition (Moody - Implicit)

$$\frac{1}{\sqrt{f}} = -2.0 \log\left[\frac{\epsilon/D}{3.7} + \frac{2.51}{(\sqrt{f})(Re)}\right]$$

A53. Reynolds number, Prandtl number, Nusselt number, and Rayleigh number:

$$Re = \frac{UL}{\nu} \; ; \; Pr = \frac{\nu}{\alpha} = \frac{\rho \nu c_p}{k} \; ; \; Nu = \frac{hL}{k} \; ; Gr = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2} \; ; \; Ra = Gr \cdot Pr$$

A6. Quasi-one-dimensional heat conduction equation:

$$\dot{Q}_x = -kA\frac{\partial T}{\partial x} \; ; \; \frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2}$$

A7. Nusselt number of force convection for flat plate:

$$Nu_L = \begin{cases} 0.66412\, Pr^{\frac{1}{3}} Re_L^{\frac{1}{2}} & Re_L \leq 5 \times 10^5 \\ Pr^{\frac{1}{3}}\left(0.037\, Re_L^{\frac{4}{5}} - 871.2\right) & Re_L \geq 5 \times 10^5 \\ 0.037\, Pr^{\frac{1}{3}} Re_L^{\frac{4}{5}} & Re_L \gg 5 \times 10^5 \end{cases}$$

A8. Nusselt number of free convection for vertical plate:

$$Nu_L = \begin{cases} \frac{4}{3}\left(\frac{Gr_L}{4}\right)^{1/4}\left[\frac{0.75\, Pr^{1/2}}{(0.609 + 1.221\, Pr^{1/2} + 1.238\, Pr)^{1/4}}\right] & Ra_L \leq 10^9 \\ \left[0.825 + 0.387\, Ra_L^{\frac{1}{6}}\left[1 + (0.492/Pr)^{\frac{9}{16}}\right]^{-8/27}\right]^2 & Ra_L > 10^9 \end{cases}$$

A9. Nusselt number of free convection for horizontal plate:

Warm surface face up / Cold surface face down

$$Nu_L = \begin{cases} 0.54\, Ra_L^{\frac{1}{4}} & 10^4 \le Ra_L \le 10^7, Pr \ge 0.7 \\ 0.15\, Ra_L^{\frac{1}{3}} & 10^7 \le Ra_L \le 10^{11}, Pr \ge 0 \end{cases}$$

Cold surface face up / Warm surface face down

$$Nu_L = \begin{cases} 0.52\, Ra_L^{\frac{1}{5}} & 10^4 \le Ra_L \le 10^9, Pr \ge 0.7 \\ 0.27\, Ra_L^{\frac{1}{4}} & 10^5 \le Ra_L \le 10^{11}, Pr \ge 0.7 \end{cases}$$

Rectangular cavities

$$Nu_L = \begin{cases} 0.22\left[\dfrac{Pr(Ra_L)}{0.2 + Pr}\right]^{0.28} (H/L)^{-\frac{1}{4}} & 10^3 \le Ra_L \le 10^{10}, Pr \le 10^5, 2 \le H/L \le 10 \\ 0.42(Ra)^{\frac{1}{4}} Pr^{0.012} (H/L)^{-0.3} & 10^4 \le Ra_L \le 10^7, 1 \le Pr \le 2 \times 10^4, 10 \le H/L \le 40 \end{cases}$$

A10. Radiation heat transfer of gray surface enclosed:

Large Parallel Planes

$$\dot{Q}_{12} = \frac{\sigma A_1 \left(T_1{}^4 - T_2{}^4\right)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$$

Long Concentric Cylinders

$$\dot{Q}_{12} = \frac{\sigma A_1 \left(T_1{}^4 - T_2{}^4\right)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2}\left(\frac{r_1}{r_2}\right)}$$

Concentric Spheres

$$\dot{Q}_{12} = \frac{\sigma A_1 \left(T_1{}^4 - T_2{}^4\right)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2}\left(\frac{r_1}{r_2}\right)^2}$$

End of Paper